7.1 Introduction

One of the important considerations for a naval architect is the powering requirement for a ship. Once the hull form has been decided upon, it is necessary to determine the amount of engine power that will enable the ship to meet its operational speed requirements.

Unfortunately, the total resistance of a ship cannot be predicted by calculation alone. Even with today's sophisticated computers and the ability to mathematically model the flow around a ship, the complex shape of a ship's hull form force the naval architect to rely upon model test data to predict ship engine size.

One goal of this chapter is to give the student an appreciation of model testing since it is such a vital part of the engineering process. The student will be taught how to calculate the effective horsepower (EHP) needed to move the hull of a ship to a given speed from model data obtained in a towing tank.

The student will be taught the physical concepts relevant to resistance and powering of ships so that the scaling technique becomes more than a rote set of equations to crunch through.

The equations listed in objective 13 will be provided for calculations in this chapter. These equations are not meant to be memorized. The student has to be able to **apply these equations, know the units of each term, and understand the physical principles** behind each equation. The equations are **listed as a matter of convenience** but will be taught in context throughout the chapter. The units for each parameter are in parenthesis to aid the understanding of each relationship. The definition of each symbol is given later in the chapter when the equation is introduced.

7.2 The Ship Drive Train

Before ship resistance and powering can be examined in any detail, the definitions and relationships between the horsepowers along the ships drive train must be quantified. Figure 7.1 shows a simplified picture of the drive train.

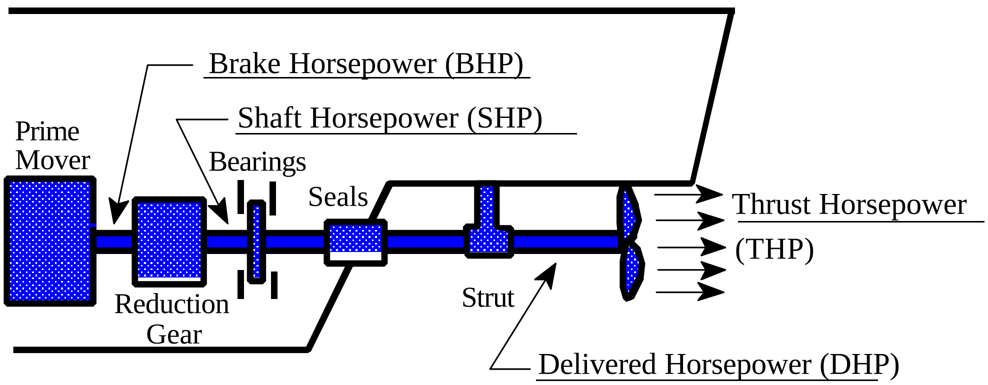


Figure 7.1 - Simplified Ship Drive Train

7.2.1 Brake Horsepower (BHP)

Brake Horsepower (BHP) is the power output at the shaft coming out of the engine before any reduction gears. Sometimes the size of the engine required to satisfy the design criteria is specified in terms of BHP but in most cases Shaft Horsepower (SHP) is used instead. The engine is considered the first element in the drive train and in most naval ships the engine will consist of a diesel engine, gas turbine, or steam turbine.

7.2.2 Shaft Horsepower (SHP)

Shaft Horsepower (SHP) is the power output after any reduction gears. Reduction gears are necessary to convert the high revolutions per minute of the engine to slower revolutions per minute required for efficient screw propeller operation. There is only a few percent loss of efficiency between BHP and SHP but SHP is always a smaller value than BHP.

NB: The reduction gears can be relatively small components as they are in some gas turbines or extremely large as they are in some steam turbines. Small and large are rather vague terms, but your instructor can reveal their meaning by taking you to the model room.

7.2.3 Delivered Horsepower (DHP)

Delivered Horsepower (DHP) is the power delivered to the propeller. It is a smaller value than the SHP due to the losses that occur in the bearings, stern tube and seals necessary to keep the water from entering the ship through the propeller shaft.

7.2.4 Thrust Horsepower (THP)

Thrust horsepower (THP) is the horsepower created by the screw propellers thrust. THP is smaller than DHP due to the inefficiencies inherent in converting rotational motion to thrust. The difference between DHP and THP represents the largest loss in the drive train and is given by a quantity called the propeller efficiency ($\eta_{propeller}$).

$$\eta_{propeller} = \frac{THP}{DHP}$$

Typically, a well designed propeller will have an efficiency of about 70%.

7.2.5 Relative Magnitudes

The relative magnitude of the powers decreases as we move from the prime mover to the screw propeller. This is because of the losses created by each element of the ship drive train. BHP has the largest magnitude, followed by the SHP, DHP and THP.

$$BHP > SHP > DHP > THP$$

From Figure 7.1 and the relationships given above, we can also relate the relative magnitudes of the powers using efficiency terms. These efficiencies are useful in defining properties of the drive train components. In addition to the propeller efficiency defined previously, we can also define the following:

$$\eta_{gear} = \frac{SHP}{BHP} \qquad \eta_{shaft} = \frac{DHP}{SHP}$$

7.3 Effective Horsepower (EHP)

Effective Horsepower (EHP) is defined as follows:

> *"The horsepower required to move the ship hull to a given speed in the absence of propeller action."*

EHP is determined from model data that is obtained from tow tank experimentation. In these tests, a model is pulled through the water by a carriage, consequently the action of the propeller is not considered.

The model test data is scaled up to find the EHP of the full scale ship using a technique called a Froude Expansion. This technique will be covered later in this chapter. By performing this procedure a number of times at different carriage speeds, the EHP for a range of ship speeds can be determined and plotted on a power curve. Figure 7.2 shows a typical ship power curve. The reason behind the shape of the power curve will be covered later, but it is clear to see that the EHP requirement for any ship speed can be found.

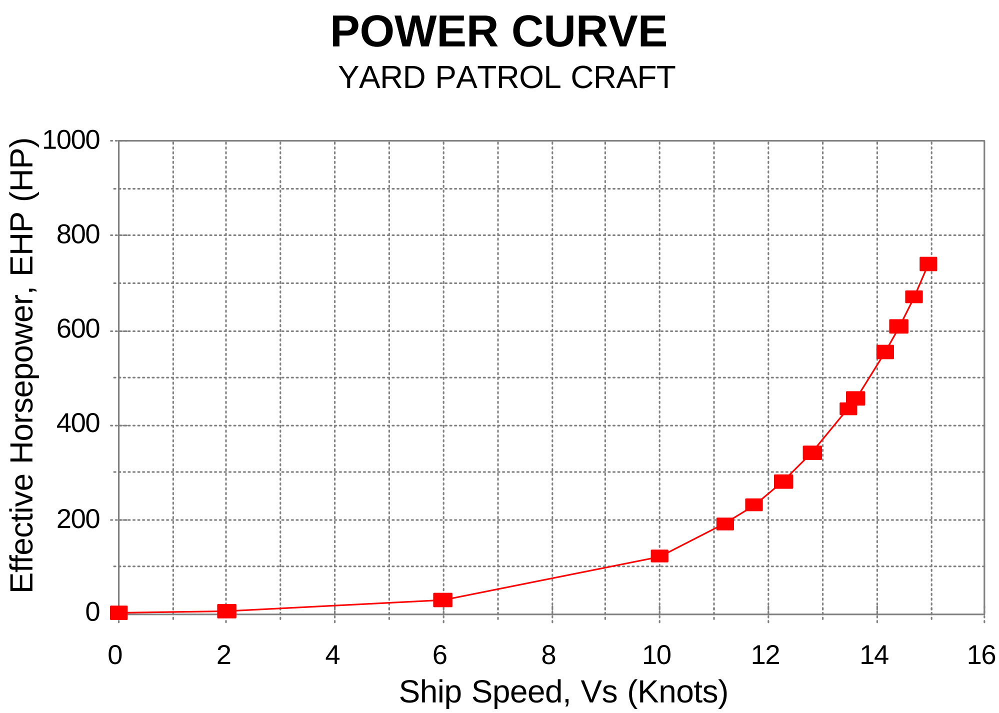


Figure 7.2 - Typical Ship Power Curve

7.3.1 Hull Efficiency

With the required EHP known to meet the operational speed requirement of the ship, it is now possible to work this value back through the various efficiencies of the drive train and determine the BHP requirement of the prime mover. However, before this is possible, we must find the relationship between EHP and the powers in the drive train.

EHP is related back to THP by a quantity called the hull efficiency (η_H) . As with other efficiencies, η_H is the ratio of EHP to THP.

$$\eta_H = \frac{EHP}{THP}$$

However, where η_H differs from other efficiencies is that it can have a value greater than one! This is due to the interaction of the hull with the propeller and the effect this interaction can have on propeller efficiency.

7.3.1.1 Hull/Propeller Interaction

The propeller efficiency ($\eta_{propeller}$) already covered is calculated for the propeller operating in clear, uninterrupted flow conditions. Clearly, when operating at the stern of a ship, this is not the case. The presence of hull appendages and the form of the hull itself can effect the flow entering the propeller race and consequently, the efficiency of the propeller.

In certain hull designs, the flow pattern created by the hull can reduce the effectiveness of the propeller, in which case the hull efficiency (η_H) will have a value < 1.

If the hull is designed well, the flow pattern created by the hull can increase the effectiveness of the propeller, in which case the hull efficiency (η_H) will have a value > 1. Values of 1.05 for η_H are not uncommon.

7.3.1.2 Calculating Hull Efficiencies (OPTIONAL)

The hull efficiency can be calculated from a knowledge of certain ship and propeller characteristics. the final equation is given here for completeness.

$$\eta_H = \frac{EHP}{THP} = \frac{[R_T]\ V_s}{T\ [V_A]} = \frac{[(1 - t)\ T]\ V_s}{T\ [(1 - w)\ V_s]} = \frac{1 - t}{1 - w}$$

where w and t are the ‘wake fraction’ and ‘thrust deduction’ respectively.

7.4 Propulsive Coefficient (PC)

Having established that the link between THP and EHP is the hull efficiency (ς_H), it is now possible to establish the BHP requirement for a ship from the magnitude of EHP obtained from the power curve. Figure 6.3 displays the block diagram of the various drive train elements and the powers at each interface which can aid this calculation.

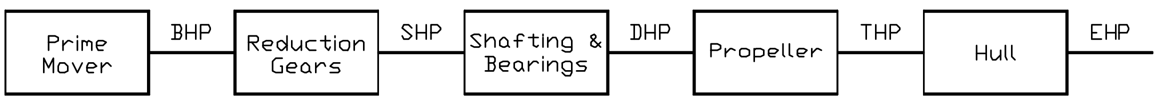


Figure 7.3 - Block Diagram of Ship Drive Train

Instead of having to deduce the effect of all the separate efficiencies down the ship drive train, the separate efficiencies are often amalgamated into one called the propulsive efficiency (ς_P) or more often the propulsive coefficient (PC). The propulsive coefficient is the ratio of EHP to SHP. Typically, a well designed propeller and drive train would produce a propulsive coefficient of about 0.6.

Provided the power curve and the propulsive coefficient for a ship are known, it is possible for the prime mover to be sized at an early stage in the ship design process.

Example 7.1 Assuming the Yard Patrol Craft has a propulsive coefficient of 0.59, calculate the SHP requirement of its prime mover if it has an operational requirement to reach 14 knots.

$$From\ Figure\ 7.2\ \ \&\ \ EHP\ @\ 14\ knots\ is\ 530\ HP$$

$$PC = \frac{EHP}{SHP}$$

$$SHP = \frac{EHP}{PC} = \frac{530\,HP}{0.59}$$

$$SHP = 900\ HP$$

7.5 Total Hull Resistance (R_T)

Total hull resistance (R_T) is the force that the ship experiences opposite to the motion of the ship as it moves through the water. The relationship between the EHP and R_T is stated below.

$$EHP(\mathrm{Hp}) = \frac{R_T(\mathrm{lb})\ V_S\left(\frac{\mathrm{ft}}{\mathrm{s}}\right)}{550\left(\frac{\mathrm{ft\ lb}}{\mathrm{s\ Hp}}\right)}$$

Where: R_T is the total hull resistance

V_S is the speed of the ship

This equation makes intuitive sense if you consider the units and the basic definitions of power from general physics. Power is the rate at which work is done or energy is expended. Work is the action of a force acting on a body through some distance. Study the equation and see if you can relate the units to the definition of power.

7.5.1 Coefficient of Total Hull Resistance (C_T)

It is convenient and a useful engineering practice to group several parameters into a dimensionless ratio and define a new quantity. Total hull resistance is often expressed in terms of a non-dimensional quantity (C_T) by dividing the total hull resistance (R_T) by the product of dynamic pressure ($0.5\rho V_S{}^2$) and wetted surface area (S). This ratio is defined to be the total coefficient of resistance (C_T) and is shown below.

$$C_T = \frac{R_T(\mathrm{lb})}{0.5\ \rho\left(\frac{\mathrm{lb}\cdot\mathrm{s}^2}{\mathrm{ft}^4}\right)\ V_S^2\left(\frac{\mathrm{ft}^2}{\mathrm{s}^2}\right)\ S(\mathrm{ft}^2)}$$

Where: C_T is the coefficient of total hull resistance in calm water

R_T is the total hull resistance

ρ is the density of the fluid

V_S is the speed of the ship

S is the wetted surface area on the submerged hull

The equation for C_T can be rearranged to solve for R_T in terms of the other parameters. Water density (ñ) is available from water property tables, the wetted surface area (S) is a geometric property determined from the curves of form, the ship's speed (V_S) is the independent variable and the coefficient of total hull resistance (C_T) can be determined by model testing. Rearranging:

$$R_T(\text{lb}) = 0.5\ \rho \left(\frac{\text{lb}\cdot\text{s}^2}{\text{ft}^4}\right)\ S(\text{ft}^2)\ V_S^2 \left(\frac{\text{ft}^2}{\text{s}^2}\right)\ C_T$$

7.5.2 Change in C_T with Increasing Speed

C_T increases with the ship speed at an increasing rate for reasons that will be discussed in the next section. For now it is important to realize that R_T is the product of both C_T and the ship's speed to the second power so that R_T is proportional to the ship's speed to the n'th power where n varies from 2 at low speeds to approximately 5 at high speeds. The plot at Figure 7.4 displays this relationship.

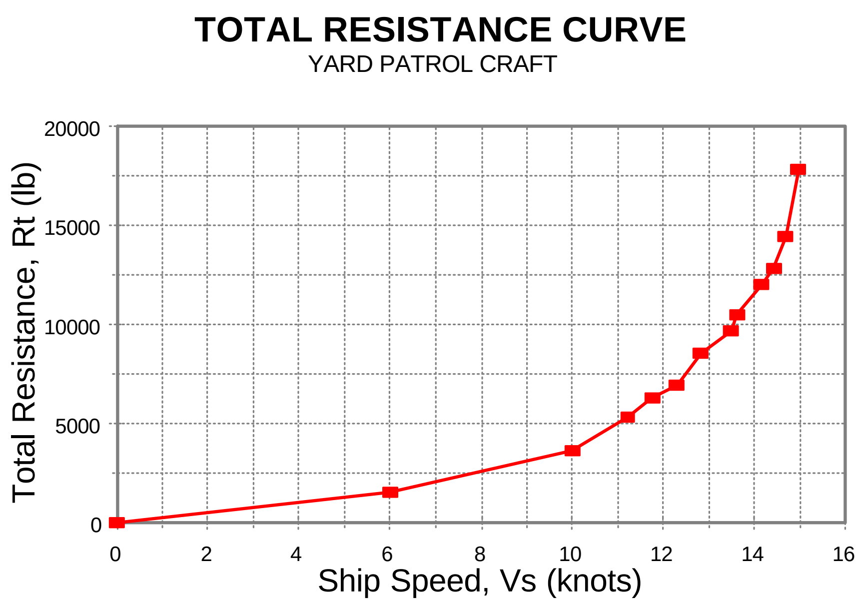


Figure 7.4 - Ship Resistance Curve

Further, EHP is the product of the total hull resistance and the speed of the ship so *n* varies from 3 at low speeds to approximately 6 at high speeds for EHP.

On a practical level, getting from point A to point B in a short time will cost an increasing amount of energy and there are practical fuel consumption limits to consider when choosing the ship's speed.

7.6 Components of Total Hull Resistance

The total resistance of a ships hull moving through calm water (R_T) can be divided into three components as shown in the next equation.

$$R_T = R_V + R_W + R_{AA}$$

Where: R_V is viscous resistance

R_W is wave making resistance

R_{AA} is resistance caused by the air

Figure 7.5 shows the variation in the magnitudes of these components with ship speed. At low speeds viscous resistance dominates; at higher speeds, the curve turns upward drastically as wave making resistance begins to dominate. Ship resistance in air is a function of the surface area of the hull and superstructure above the waterline.

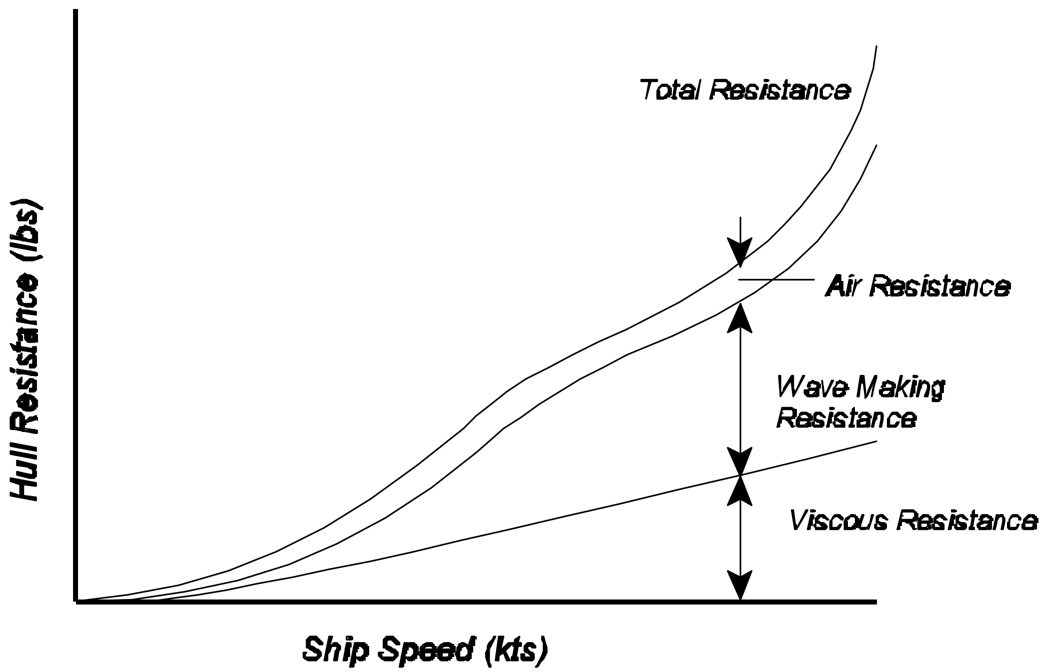


Figure 7.5 - Variation in Components of Ship Resistance with Ship Speed

The “hump” that occurs on the total resistance curve is a wave making effect. The location of the hump is a function of ship length and ship speed that will be explained later.

7.6.1 Viscous Resistance (R_V) and the Coefficient of Viscous Resistance (C_V)

Viscous Resistance (R_V) accounts for the resistance from the viscous stresses that the fluid exerts on the hull. Viscous resistance can also be considered in terms of a dimensionless coefficient, (C_V).

Viscosity is a property of the fluid that describes how much resistance a fluid has to flow. Syrup is said to be very viscous; the fluid particles offer a lot of resistance to other fluid particles and to bodies in the flow. On the other hand, alcohol has a low viscosity with little interaction between fluid particles and it flows more easily.

There are 2 viscous components that make up C_V, one is tangential to the direction of flow, the other is normal (or perpendicular) to the flow.

7.6.1.1 The Tangential Component (C_F)

The tangential component yields stresses that are parallel to the ship's hull. As the ship moves through the water, fluid friction acting over the entire wetted surface causes a net force opposing motion. In naval architecture, this component is called the 'skin friction' which itself can be written in a non-dimensional form called the Coefficient of Skin Friction (C_F).

Values of C_F are obtained from curve fits of experimental data. The experiments are done on flat plates that are pulled through the water so that only the shearing component is contributing. Over the years many empirical equations have been developed to match these experimental results but in 1957 the International Towing Tank Conference (ITTC) agreed the use of the following semi-empirical equation which is a function of the Reynolds number (R_n).

$$\textit{Tangential Component of } C_V = C_F = \frac{0.075}{\left(\log_{10} R_n - 2\right)^2}$$

NB: Whenever scientists fit experimental data with an equation the equation is said to be empirical. That is, it is not based on theory. If some theory is used to develop a partial expression but the expression must be altered to fit the data, then the equation is called a semi-empirical equation.

NB: The Reynolds number (R_n) is a dimensionless grouping of fluid flow parameters, the ratio of length multiplied by speed over the kinematic viscosity. It is named after Sir Osborne Reynolds (1883) in honor of his accomplishments in the study of fluid flow and is a very useful and convenient quantity in the study of fluids.

The equation for Reynolds Number (R_n) is given below.

$$R_n = \frac{L(\text{ft})\ V_S\left(\frac{\text{ft}}{\text{s}}\right)}{\nu\left(\frac{\text{ft}^2}{\text{s}}\right)}$$

Where: R_n is the Reynolds Number

L is the L_{PP}

V_S is the speed of the ship

ν is the kinematic viscosity of the water

The magnitude of R_n indicates the type of flow pattern being experienced by an object.

The flow of fluids can be roughly divided into laminar and turbulent flow. Laminar flow is the condition where the fluid flows in layers in an orderly fashion. Turbulent flow is the condition where the flow is chaotic and well mixed. As the water flows from the bow to the stern the flow undergoes a transition from laminar to turbulent to flow separation. All three flow types are shown in Figure 7.6 which is reproduced from “Introduction to Naval Architecture” by Gillmer and Johnson.

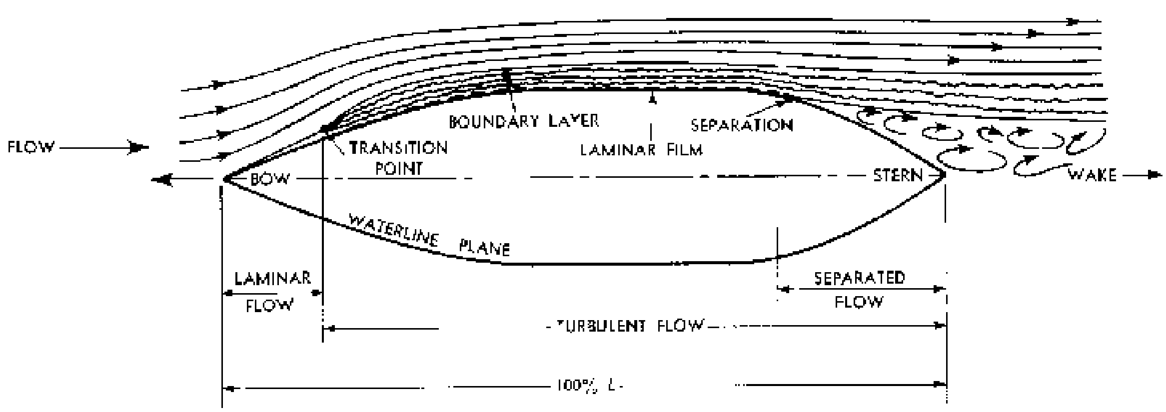


Figure 7.6 - The Fluid Flow Pattern Surrounding a Ship

For external flow over flat plates, typical Reynolds Number (R_n) magnitudes are as follows:

Laminar Flow - $R_n <$ about 5×10^5

Turbulent Flow - R_n > about 1 X 10^6

Values between these numbers is the transition zone from laminar to turbulent.

NB: Turbulence and flow separation represent greater viscous resistance as compared to laminar flow. A real ship moves mostly in turbulent flow even at very slow speeds. Only the first few feet of the bow experience laminar flow. This raises an issue when model testing. The models used in tow tanks experiments also should experience a turbulent flow as they are towed by the carriage. The model surface appears must smoother than a ships and the flow may not be turbulent. To insure each model experiences turbulent flow small studs are attached vertically on the sides of the model. These studs stick out and agitate the water inducing turbulence.

7.6.1.2 The Normal Component

The normal component of viscosity causes a pressure distribution along the underwater hull form of the ship. A high pressure is formed in the forward direction opposing motion and a low pressure is formed aft. The low pressure results in eddy currents and water being dragged along with the ship.

The magnitude of this normal component is obviously effected by the form or shape of the hull. Larger, fuller hull forms will have a greater normal component of viscosity than slender hull forms. Consequently it is no surprise that the normal component of viscous resistance is calculated by the product of the Coefficient of Skin Friction (C_F) with a value called the Form Factor (K).

$$Normal\ Component\ of\ C_V = K\,C_F$$

Where: C_F is the Coefficient of Skin Friction

K is the Form Factor

$$K = 19\left(\frac{\nabla(\mathrm{ft}^3)}{L(\mathrm{ft})\,B(\mathrm{ft})\,T(\mathrm{ft})}\,\frac{B(\mathrm{ft})}{L(\mathrm{ft})}\right)^2$$

7.6.1.3 Calculating the Coefficient of Viscous Resistance C_V

The viscous resistance coefficient (C_V) is expressed as the sum of the 2 terms already discussed, the tangential component of viscosity and the normal component. The equation is given below.

$$C_V = Tangential\ Component + Normal\ Component$$

$$C_V = C_F + K\,C_F$$

Where: $$K = 19\left(\frac{\nabla(\mathrm{ft}^3)}{L(\mathrm{ft})\,B(\mathrm{ft})\,T(\mathrm{ft})}\,\frac{B(\mathrm{ft})}{L(\mathrm{ft})}\right)^2$$

$$C_F = \frac{0.075}{\left(\log_{10} R_n - 2\right)^2} \;.\;.\;.\;.\;.\;.\;.\;.\;.\; R_n = \frac{L\,(ft)\;V_S\left(\frac{ft}{s}\right)}{\nu\left(\frac{\mathrm{ft}^2}{\mathrm{s}}\right)}$$

Clearly, with the geometry of a ship known, the Coefficient of Viscous Resistance is easily computed using these equations.

7.6.1.4 Reducing the Coefficient of Viscous Resistance C_V

An analysis of the equations above clearly relate the advantages of increasing a ships length (L) in reducing the magnitude of C_V.

- Increasing L but maintaining a ship's underwater volume will reduce the value of the Form Factor (K). Hence the normal component of viscous resistance will reduce. This is an obvious observation - a slender hull form will create a smaller pressure differential between the bow and stern regions.

- Increasing L will increase the magnitude of the Reynolds Number (R_n) for a particular ship speed. Following this into the ITTC, 1957 equation for C_F reveals a reduction in the value for C_F. This then reduces the tangential component and normal component of C_V.

NB: The increase in L must be achieved without an increase in ship displacement or submerged volume. Hence this increase must coincide with a decrease in ship depth or breadth. This is not always desirable. The improvements in ship resistance offered by the length increase must be evaluated against the lower level of ship stability, survivability and carrying capacity produced by such a change.

7.6.2 The Coefficient of Wave Making Resistance (C_W)

C_W is the coefficient of wave making resistance. A body traveling at or near the surface will create waves. The creation of wave requires energy. Any energy used for making waves represents lost energy which could have been used to make the ship go faster. Wave making resistance is much less at slow speeds, but becomes the major component of resistance at high speeds.

Two types of wave patterns are created by a ship moving through the water. They are the divergent and transverse wave systems which are illustrated below at Figure 7.7. This figure is reproduced from “Introduction to Naval Architecture” by Gillmer and Johnson.

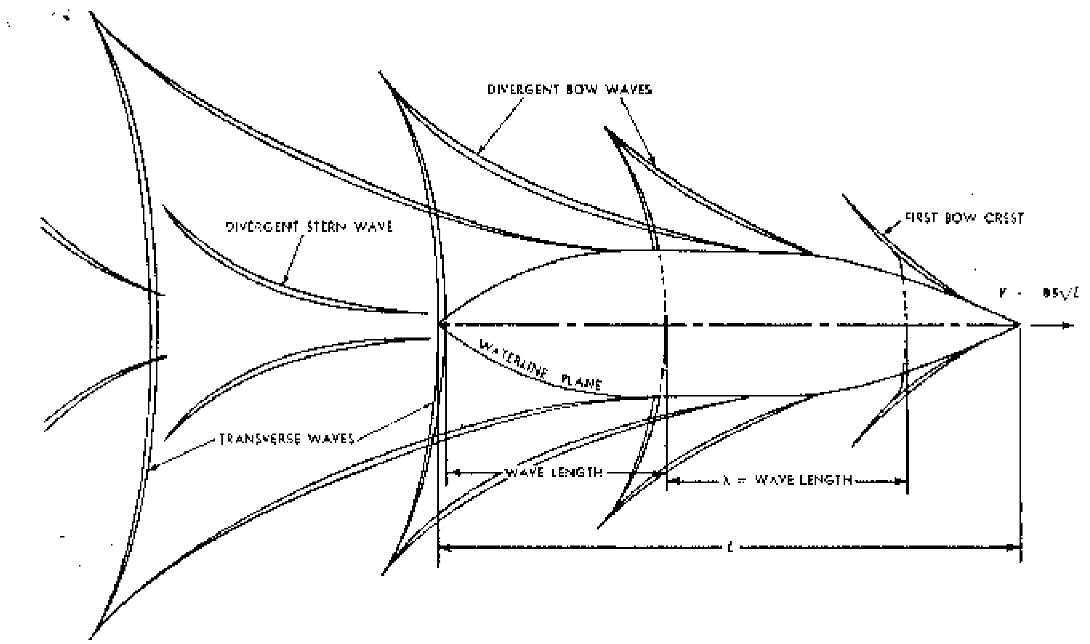


Figure 7.7 - The Divergent and Transverse Wave Pattern Generated by a Ship

7.6.2.1 Transverse Wave System

The transverse wave system travels at approximately the same speed as the ship. This is by necessity since the ship is producing the wave. At slow speeds, the waves are short and several crests are seen along the ship's length. As the ship speeds up, the length of the transverse wave increases. When the length of the transverse wave approaches the length of the ship the wave making resistance increases very rapidly. The ship in effect must power itself through this wave if it is to go any faster. At this point the energy expenditure increases much more rapid than the increase in speed. This is the main reason for the dramatic increase in R_T as speed increase as indicated at Figure 7.5.

The speed at which the transverse wave created by the ship equals the length of the ship is referred to as “hull speed”. There is nothing magic about the term hull speed, it merely describes a physical situation that would other wise take more words to explain.

Going faster than the hull speed results in a transverse wave longer than the ship, however the crest of the wave is still attached to bow since the ship is creating the wave. At yet faster speeds the ship stops behaving like a displacement ship (one that floats due to Archimedes Principle) and starts behaving like a planing craft.

7.6.2.2 The Divergent Wave System

The divergent wave system consists of the bow and stern waves. Some hull forms will generate another divergent system from close to their mid-body.

The interaction of the bow and stern waves can explain the ‘hump phenomena’ observed on the ship resistance curve at Figure 7.5. As the ship increases speed, the phase difference between the bow and stern wave systems alters. At some speeds the waves will be in phase (peak matches peak) at others they will be out of phase (peak matches trough).

When the waves are in phase, the addition of the 2 peaks creates a large divergent wave system. This phenomena is called constructive interference. Bigger waves take more energy to create resulting in a “hump” in the resistance curve. When the waves are out of phase, the addition of a peak with a trough creates a small divergent wave system. This phenomena is called destructive interference. Smaller waves take less energy to create resulting in a “hollow” in the resistance curve.

7.6.2.3 Calculating the Coefficient of Wave Making Resistance (C_W)

It is known that wave making resistance is a function of the Beam to Length ratio, the displacement, the shape of the hull and a dimensionless quantity called the Froude Number (F_n). This quantity will be explained in the next section. Unfortunately, the calculation of C_W has proved far too difficult and inaccurate from any theoretical, semi-empirical or empirical equation. This is despite modern computer power, and the ability to mathematically model the fluid flow around an object. The problem arises in the interaction of the highly complicated shape of the hull with the fluid at the fluid/air boundary. The calculation of C_W must rely upon model tow tank testing and in particular a technique called a Froude Expansion. This will be covered at some depth in the next section.

NB: This problem is much the same in other engineering fields. For example: calculating the drag experienced on an aircraft or automobile is only possible with the aid modeling data from wind tunnel tests.

7.6.2.4 Reducing the Coefficient of Wave Making Resistance (C_W)

! **Length** As with C_V, a greater length will help lower the wave making resistance. Hull speed for an FFG7 (L = 408 ft) is about 27 knots. This speed places FFG7 in the region of the resistance curve which slopes rapidly upward. In this region every extra knot of speed costs a great number of horsepower. Hull speed for CVN65, *Enterprise* (L = 1040 ft), is about 43 knots. At 30 knots, *Enterprise* is still in the very flat region of its resistance curve, where each extra knot costs very little in horsepower.

It would be very difficult to add enough propulsion machinery to increase FFG7's speed significantly above hull speed. If it is desired for the frigate to travel with a battle group, which it is, then the ships are given enough power to do about the same speed. Because of increased wave making resistance at speeds above hull speed, longer ships use proportionally smaller engines to do the same speed. In other words, it requires fewer horsepower per ton to make CVN65 do 30 knots than it does to make FFG7 do 30 knots.

NB: One of the nautical myths floating around is the possibility of a 60 knot aircraft carrier. For carriers of current dimensions, 60 knots is far above hull speed. There is no way one could provide enough propulsion to overcome the wave resistance that far above hull speed. If it were desired to build an aircraft carrier whose hull speed was about 60 knots, it would need to be about 2700 ft long. Considering that the top speed of current aircraft carriers is well below hull speed, a more reasonable guess would place such a ship at over three times as long as current aircraft carriers. Just try to get that funded....

! **Bulbous Bows** Bulbous Bows are one attempt to reduce the wave making resistance of surface ships by reducing the divergent wave system. The bulbous bow was developed by RADM David Taylor and was used as early as 1907 on the battleship *USS Delaware*. The idea behind a bulbous bow is to create a second bow wave which interferes destructively with the bow wave normally seen, ideally resulting in no bow wave, in reality resulting in a smaller bow wave. A smaller wave takes less energy away from the objective - making the ship go fast. Many warships carry sonar equipment in bow bulbs.

7.6.3 Correlation Allowance

C_A is the correlation allowance and accounts for hull resistance due to surface roughness, paint roughness, corrosion, and fouling of the hull surface. With corrosion and fouling by marine life the frictional resistance can increase by as much as 50%.

The correlation allowance is dependent on the vessel being modeled and the condition of the hull when it is modeled, and is only used when a full-scale prediction of EHP is being made from model test results. Models which are towed in the tow tank have a correlation allowance equal to zero since the hull form is considered perfectly smooth. There are several empirical formulas that can be used to predict the correlation allowance for full scale ships. None are given in this course, therefore you will always be given a value for C_A.

7.6.4 Other Types of Resistance Not Included in Total Hull Resistance (R_T)

In addition to the major sources of resistance mentioned above, there are several others that can influence the resistance experienced by a ship.

7.6.4.1 Appendage Resistance

Appendage Resistance is the drag caused by all of the underwater appendages , including the propeller shafting, bilge keels, struts, and rudders. In Naval ships, appendages can account for approximately 2 - 14% of the total resistance.

If the model is large enough hull appendages such as rudder, skeg, and bilge keel will be put on the model. In these circumstances, appendage resistance will be included in the value of R_T.

If the model is small then the appendages are left off and just a bare hull is used but this results in underestimating the EHP. It is then a common procedure to add a percentage to this value to account for appendages. This percentage will depend upon the type and configuration of the hull being tested.

NB: A larger model is preferred so appendages can be included. Additionally, any errors in the testing will be multiplied by the geometric scale factor between the ship and model. The larger model will have a smaller geometric scale factor and result in less magnification of any errors.

7.6.4.2 Steering Resistance

Steering Resistance is caused by the motion of the rudder. Every time the rudder is moved, additional drag is created. Although only a small component of the resistance of a warship, steering resistance can be particularly troublesome in sailboats.

7.6.4.3 Air and Wind Resistance (R_{AA})

Air Resistance is the resistance caused by the flow of air over the ship with no wind present. This component is affected by the area and shape of the ship above the waterline, wind velocity and wind direction. Resistance from air and wind is typically 4 - 8% of the total ship resistance, but may be as much as 10% in high sided ships such as aircraft carriers. Wind resistance is an additional component of resistance which is a function of wind speed. At a wind speed of 20 knots, ship's resistance may be increased by up to 25 - 30%.

7.6.4.4 Added Resistance due to Waves

NB: Added resistance due to waves is not to be confused with wave making resistance.

Added resistance due to waves refers to ocean waves caused by wind and storms: waves which would be present without the ship. The waves cause the ship to expend energy by pitching, rolling, yawing and diffracting waves. Again, motion produced in any direction other than straight forward represents wasted energy. This component of resistance can be very significant in high sea states.

7.6.4.5 Increased Resistance in Shallow Water

Increased resistance in shallow water is caused by several effects.

- The flow around the bottom of the hull is restricted in shallow water, therefore the water flowing under the hull speeds up. The faster moving water increases the frictional resistance.
- The faster moving water lowers the pressure under the hull, sucking more of the ship into the water, increasing wetted surface area and increasing frictional resistance.
- The waves produced in shallow water take more energy from the ship than they do in deep water for the same speed. Thus, wave making resistance also increases in shallow water.

The net result is that it takes more horsepower (and therefore gas) to make your SOA in shallow water that in deep water.

7.7 Basic Theory Behind Ship Modeling

As discussed earlier, a vessel is modeled because it is the only way, short of building and testing a full scale prototype, of determining the total resistance. In order for the results obtained from the model to predict the real ship, 2 fundamental conditions should be met. The conditions of geometric and dynamic similarity.

7.7.1 Geometric Similarity

Geometric similarity is obtained when all characteristic dimensions of the model are directly proportional to the ships dimensions. The model is then a scaled version of the real ship. The ratio of the length of the ship to the length of the model is typically used to define the scaling factor **(λ)**. However, any characteristic length ratio's between the ship and the model could be used.

$$\text{Scale factor} = \lambda = \frac{L_S(ft)}{L_M(ft)}$$

From this it follows logically the ratio of areas should be equal to the scale factor squared and ratios of volumes should be equal to the scale factor cubed. The characteristic area of importance for modeling is the wetted surface area of the underwater hull (S). The characteristic volume of importance is the submerged hull volume (∇). These relationships are shown below.

$$\lambda^2 = \frac{S_S(ft^2)}{S_M(ft^2)} \qquad\qquad \lambda^3 = \frac{\nabla_S(ft^3)}{\nabla_M(ft^3)}$$

NB: In the equations above and all the equations that follow, quantities associated with geometry and resistance will have a suffix of either “S” or “M”.

- The suffix “S” refers to a quantity for the full scale ship.
- The suffix “M” refers to a quantity for the model.

7.7.2 Dynamic Similarity

Dynamic similarity requires that the forces associated with the fluid motions around both the model and ship have scaled magnitudes and identical directions at corresponding locations. The model must behave in exactly the same way as the real ship.

One way to assure dynamic similarity is to run the model tests such that the relevant dimensionless parameter groupings are forced to be equal for the model and ship. So far we have seen that the 2 major components of the coefficient of total resistance (C_T) is the coefficient of viscous resistance (C_V) and the coefficient of wave making resistance (C_W). These are functions of the 2 dimensionless parameters Reynolds Number (R_n) and Froude Number (F_n) respectively.

$$C_V = f(R_n) \qquad\qquad C_W = f(F_n)$$

Consequently, by making both the Froude Numbers and Reynolds Numbers the same for the model and the ship, the model will behave in exactly the same manner as the real ship. Dynamic similarity will have been achieved.

Unfortunately this is physically impossible. It is impossible to set up model tests in such a manner due to the presence of constants such as the acceleration due to gravity (g) and the kinematic viscosity of water (ν) contained in both R_n and F_n. Gravity and water properties cannot be scaled. So a choice has to be made.

We have seen that there is a convenient semi-empirical equation that can be used to calculate the coefficient of viscosity (C_V) dependant upon the geometric characteristics of the hull shape. However, there is no such relationship for the wave making coefficient (C_W).

Consequently, a conscience choice is made to make F_n the same for the model and ship and to have R_n different. This is known as incomplete or partial dynamic similarity. In tow tank testing, partial dynamic similarity is achieved by towing the model at speeds that adhere to the "Law of Comparison" sometimes called the "Law of Corresponding Speeds".

7.7.3 The Law of Corresponding Speeds

William Froude determined the "Law of Corresponding Speeds". He noticed that the wave patterns of the model looked the same as the wave pattern of the ship when the model and ship were traveling at the same speed to square root of length ratio.

The Law of Corresponding Speeds

$$\frac{V_S\left(\frac{\text{ft}}{\text{s}}\right)}{\sqrt{L_S\,(\text{ft})}} = \frac{V_M\left(\frac{\text{ft}}{\text{s}}\right)}{\sqrt{L_M\,(\text{ft})}}$$

Because the wave patterns of the model and ship were similar using this relationship, Froude decided that it would be correct to use the same value of wave making coefficient (C_W) for the model and ship when operating under these conditions.

This is the key concept that allows naval architects to use the model data to find the hull resistance of the real ship.

The ratio of speed to square root of length was made a dimensionless quantity by multiplying the denominator by the square root of the acceleration of gravity. This quantity was called the Froude number (F_n) to honor the work of William Froude (1810-1878) in modeling wave resistance.

$$\text{Froude number} = F_n = \frac{V_S\left(\frac{\text{ft}}{\text{s}}\right)}{\sqrt{g\left(\frac{\text{ft}}{\text{s}^2}\right) L(\text{ft})}}$$

7.7.4 Modeling Theory Summary

The coefficient of viscous resistance (C_V) is a function of Reynolds number and is found with semi-empirical equations. C_V will be different for the model and ship since the Reynolds number cannot be made to be equal for the model and ship when using the "law of corresponding speeds". C_V is the sum of the tangential viscous forces called the skin friction (C_F) and the normal viscous forces which is the product of C_F and the form factor (K) to account for the shape of the hull.

NB: Note that the form factor (K) will be the same for the model and ship since it is a ratio of geometric parameters that are proportional by the same scale factor (ë).

The coefficient of wave making resistance (C_W) is a function of the Froude number which is forced to be the same for the model and ship by using the "law of corresponding speeds". Finding the value of C_W involves the performance of a Froude Expansion as explained in the next section.

The correlation allowance (C_A) is given by an empirical equation (for this class a value will be given) for the ship or is equal to zero for the model.

Knowing all three coefficients for the ship gives the coefficient of total hull resistance of the ship (C_T) since it is the sum of all 3.

Once C_T is known, R_T and EHP can be calculated.

Example 7.2 If a model has a scale factor of 20 with a ship, what speed in kts would it need to be towed to model the ship moving at 20 kts according to Froude's law of corresponding speeds?

$$V_M = V_S\sqrt{\frac{L_M}{L_S}} = V_S\,\frac{1}{\sqrt{\text{ë}}} = 20\,kts\,\frac{1}{\sqrt{20}} = 4.47kts$$

7.8 The Froude Expansion - The Procedural Steps to Calculate EHP of a Ship from Tow Tank Data.

Model Calculations

1. Measure the length of the model and ship to obtain the scaling factor (λ).

$$\text{Scale factor} = \lambda = \frac{L_S(\text{ft})}{L_M(\text{ft})}$$

2. Choose a ship speed to model and calculate the corresponding speed at which to run the model.

$$\frac{V_S(\text{ft/s})}{\sqrt{L_S(\text{ft})}} = \frac{V_M(\text{ft/s})}{\sqrt{L_M(\text{ft})}}$$

3. Tow the model in the tow tank at the corresponding speed and measure the pounds of resistance ($R_{T\,M}$)

4. Look up the density of the water in the towing tank (ρ_M).

5. Calculate the wetted surface area of the model (S_M)

$$\lambda^2 = \frac{S_S\,(\text{ft}^2)}{S_M\,(\text{ft}^2)}$$

6. Calculate the coefficient of total hull friction for the model ($C_{T\,M}$) using the values you found in 4 & 5.

$$C_{T\,M} = \frac{R_{T\,M}(\text{lb})}{0.5\ \rho_M(\text{lb}\cdot\text{s}^2/\text{ft}^4)\ V_M^2(\text{ft}^2/\text{s}^2)\ S_M(\text{ft}^2)}$$

7. Calculate the Reynolds number of the model ($R_{n\,M}$).

$$R_{n\,M} = \frac{L_M(\text{ft})\ V_M(\text{ft/s})}{\nu_M(\text{ft}^2/\text{s})}$$

8. Calculate the form coefficient (K) for the model which will be the same for the ship.

$$K = 19\left(\frac{\nabla(\text{ft}^3)}{L(\text{ft})B(\text{ft})T(\text{ft})}\,\frac{B(\text{ft})}{L(\text{ft})}\right)^2$$

NB: Use all model measurements or all ship measurements but don't mix and match!

9. Calculate the coefficient of friction of the model ($C_{F\,M}$)

$$C_{F\,M} = \frac{0.075}{\left(\log_{10} R_{n\,M} - 2\right)^2}$$

10. Calculate the coefficient viscous resistance of the model ($C_{V\,M}$)

$$C_{V\,M} = C_{F\,M}\,(1 + K)$$

11. Calculate the coefficient of wave making resistance of the model ($C_{W\,M}$). Remember that C_{AM} is zero since the model is considered to be smooth.

Solve this equation for C_{WM}

$$C_{T\,M} = C_{V\,M} + C_{WM} + C_{A\,M}$$

Ship Calculations

12. Calculate the Reynolds number of the ship ($R_{n\,S}$).

$$R_{n\,S} = \frac{L_S(\text{ft})\;V_S(\text{ft/s})}{\nu_S(\text{ft}^2/\text{s})}$$

13. Calculate the coefficient of friction of the ship ($C_{F\,S}$).

$$C_{F\,S} = \frac{0.075}{\left(\log_{10} R_{n\,S} - 2\right)^2}$$

14. Calculate the viscous coefficient of friction of the ship ($C_{V\,S}$)

$$C_{VS} = C_{FS}\,(1 + K)$$

15. Calculate the coefficient of total hull resistance of the ship ($C_{T\,S}$). Recall that the coefficient of wave making resistance for the model ($C_{W\,M}$) is the same as for the ship ($C_{W\,S}$) when the model and ship are at corresponding speeds.

$$C_{TS} = C_{VS} + C_{WS} + C_{AS}$$

16. Calculate the total hull resistance of the ship ($R_{T\,S}$).

$$R_{TS}(\text{lb}) = 0.5\;\rho_S\,(\text{lb}\cdot\text{s}^2/\text{ft}^4)\;S_S\,(\text{ft}^2)\;V_S^2\,(\text{ft}^2/\text{s}^2)\;C_{TS}$$

17. Calculate the effective horsepower that corresponds to the corresponding ships speed for which the model test was run.

$$EHP\,(\text{Hp}) = \frac{R_{TS}(\text{lb})\;V_S\,(\text{ft/s})}{550\,(\text{ft}\cdot\text{lb/s}\cdot\text{Hp})}$$

18. Only one point on the speed - power curve is obtained. The experiment and expansion is repeated at other model speeds in order to plot the entire power curve.

Example 7.3 Given the following information, perform a Froude Expansion to calculate the EHP of the ship at this speed.

Parameter	Ship	Model
Length, L (ft)	350	5.5
Displacement, Δ (LT)	2,720	Unknown
Wetted Surface Area, S (ft^2)	15,797	Unknown
Kinematic Viscosity, ν (ft^2/s)	1.2791×10^{-5}	1.066×10^{-5}
Density, ρ ($lb\text{-}s^2/ft^4$)	1.9905	1.9365
Velocity, V (ft/s)	Unknown	5.55
Total Resistance, R_T (lb)	Unknown	0.845
Correlation Allowance, C_A	0.00025	0.0000
Form Factor, K	0.25	0.25

Model Calculations

1. Measure the length of the model and ship to obtain the scaling factor (λ).

$$\text{Scale factor} = \lambda = \frac{L_S(\text{ft})}{L_M(\text{ft})} = \frac{350\,\text{ft}}{5.5\,\text{ft}} = 63.636$$

2. Choose a ship speed to model and calculate the corresponding speed at which to run the model.

$$\frac{V_S(\text{ft/s})}{\sqrt{L_S(\text{ft})}} = \frac{V_M(\text{ft/s})}{\sqrt{L_M(\text{ft})}}$$

$$V_S(\text{ft/s}) = V_M(\text{ft/s})\,\frac{\sqrt{L_S(\text{ft})}}{\sqrt{L_M(\text{ft})}} = V_M(\text{ft/s})\,\sqrt{\lambda}$$

$$V_S(\text{ft/s}) = 5.55\,\text{ft/s}\,\sqrt{63.636} = 44.27\,\text{ft/s}$$

3. Tow the model in the tow tank at the corresponding speed and measure the pounds of resistance ($R_{T\,M}$)

Value given in table

4. Look up the density of the water in the towing tank (ρ_M).

Value given in table

5. Calculate the wetted surface area of the model (S_M)

$$\lambda^2 = \frac{S_S\,(\text{ft}^2)}{S_M\,(\text{ft}^2)}$$

$$S_M\,(\text{ft}^2) = \frac{S_S\,(\text{ft}^2)}{\lambda^2} = \frac{15{,}797\,\text{ft}^2}{63.636^2} = 3.90\,\text{ft}^2$$

6. Calculate the coefficient of total hull friction for the model ($C_{T\,M}$) using the values you found in 4 & 5.

$$C_{TM} = \frac{R_{TM}(\text{lb})}{0.5\;\rho_M(\text{lb-s}^2/\text{ft}^4)\;V_M^2(\text{ft}^2/\text{s}^2)\;S_M(\text{ft}^2)}$$

$$C_{TM} = \frac{0.845\,\text{lb}}{(0.5)\;(1.9365\,\text{lb-s}^2/\text{ft}^4)\;(5.55\,\text{ft}^2/\text{s})^2\;(3.90\,\text{ft}^2)} = 7.265\ \text{x}\ 10^{-3}$$

7. Calculate the Reynolds number of the model ($R_{n\,M}$).

$$R_{nM} = \frac{L_M(\text{ft})\;V_M(\text{ft/s})}{\nu_M(\text{ft}^2/\text{s})}$$

$$R_{nM} = \frac{5.5\,\text{ft}\;\;5.55\,\text{ft/s}}{1.066\ \text{x}\ 10^{-5}\,\text{ft}^2/\text{s}} = 2.863\ \text{x}\ 10^{6}$$

8. Calculate the form coefficient (K) for the model which will be the same for the ship.

Value given in table

9. Calculate the coefficient of friction of the model ($C_{F\,M}$)

$$C_{F\,M} = \frac{0.075}{\left(\log_{10} R_{n\,M} - 2\right)^2}$$

$$C_{F\,M} = \frac{0.075}{\left(\log_{10} 2.863 \text{ x } 10^{6} - 2\right)^2} = 3.776 \text{ x } 10^{-3}$$

10. Calculate the coefficient viscous resistance of the model ($C_{V\,M}$)

$$C_{V\,M} = C_{F\,M}\,(1 + K)$$

$$C_{V\,M} = 3.776 \text{ x } 10^{-3}\,(1 + 0.25) = 4.72 \text{ x } 10^{-3}$$

11. Calculate the coefficient of wave making resistance of the model ($C_{W\,M}$). Remember that C_{AM} is zero since the model is considered to be smooth.

$$C_{T\,M} = C_{V\,M} + C_{WM} + C_{A\,M}$$

$$C_{WM} = C_{T\,M} - C_{V\,M}$$

$$C_{WM} = 7.265 \text{ x } 10^{-3} - 4.72 \text{ x } 10^{-3} = 2.545 \text{ x } 10^{-3}$$

Ship Calculations

12. Calculate the Reynolds number of the ship ($R_{n\,S}$).

$$R_{n\,S} = \frac{L_S\,(\text{ft})\;\; V_S\,(\text{ft/s})}{\nu_S\,(\text{ft}^2/\text{s})}$$

$$R_{n\,S} = \frac{350\,\text{ft}\;\; 44.27\,\text{ft/s}}{1.2791 \text{ x } 10^{-5}\,\text{ft}^2/\text{s}} = 1.21 \text{ x } 10^{9}$$

13. Calculate the coefficient of friction of the ship ($C_{F\,S}$).

$$C_{FS} = \frac{0.075}{\left(\log_{10} R_{n\,S} - 2\right)^2}$$

$$C_{FS} = \frac{0.075}{\left(\log_{10} 1.21 \text{ x } 10^{9} - 2\right)^2} = 1.49 \text{ x } 10^{-3}$$

14. Calculate the viscous coefficient of friction of the ship ($C_{V\,S}$)

$$C_{VS} = C_{FS}\,(1 + K)$$

$$C_{VS} = 1.49 \text{ x } 10^{-3}\,(1 + 0.25) = 1.868 \text{ x } 10^{-3}$$

15. Calculate the coefficient of total hull resistance of the ship ($C_{T\,S}$). Recall that the coefficient of wave making resistance for the model ($C_{W\,M}$) is the same as for the ship ($C_{W\,S}$) when the model and ship are at corresponding speeds.

$$C_{TS} = C_{VS} + C_{WS} + C_{AS}$$

$$C_{TS} = 1.868 \text{ x } 10^{-3} + 2.545 \text{ x } 10^{-3} + 0.00025$$

$$C_{TS} = 4.66 \text{ x } 10^{-3}$$

16. Calculate the total hull resistance of the ship ($R_{T\,S}$).

$$R_{TS}(\text{lb}) = 0.5\ \rho_S(\text{lb-s}^2/\text{ft}^4)\ S_S(\text{ft}^2)\ V_S^2(\text{ft}^2/\text{s}^2)\ C_{TS}$$

$$R_{TS}(\text{lb}) = (0.5)\ (1.9905\,\text{lb-s}^2/\text{ft}^4)\ (15{,}797\,\text{ft}^2)\ (44.27\,\text{ft/s})^2\ (4.66 \text{ x } 10^{-3})$$

$$R_{TS}(\text{lb}) = 1.437 \text{ x } 10^{5}\,\text{lb}$$

17. Calculate the effective horsepower that corresponds to the corresponding ships speed for which the model test was run.

$$EHP(\text{Hp}) = \frac{R_{TS}(\text{lb})\ V_S(\text{ft/s})}{550\,(\text{ft-lb/s-Hp})} = \frac{1.437 \text{ x } 10^{5}\,\text{lb}\ \ 44.27\,\text{ft/s}}{550\,\text{ft-lb/s-Hp}}$$

$$EHP(\text{Hp}) = 11{,}570\ \text{Hp}$$

7.9 The Screw Propeller

The theory behind the design of screw propellers is very complicated and a whole subject in itself. However, there are a few definitions and screw propeller characteristics that should be known.

7.9.1 Screw Propeller Definitions

***Diameter* (D)** Twice the distance from the propeller axis to the blade tip.

Hub The connection between the blades and the propeller shaft.

Blade Tip The furthest point on the blade from the propeller hub.

Blade Root The point where the blade meets the propeller hub.

***Pitch* (P)** The horizontal distance that would be traveled ideally if the screw advanced one revolution like a sheet metal screw. Figure 7.8 reproduced from an "Introduction to Naval Architecture" by Gillmer and Johnson shows the pitch of a fixed pitch propeller.

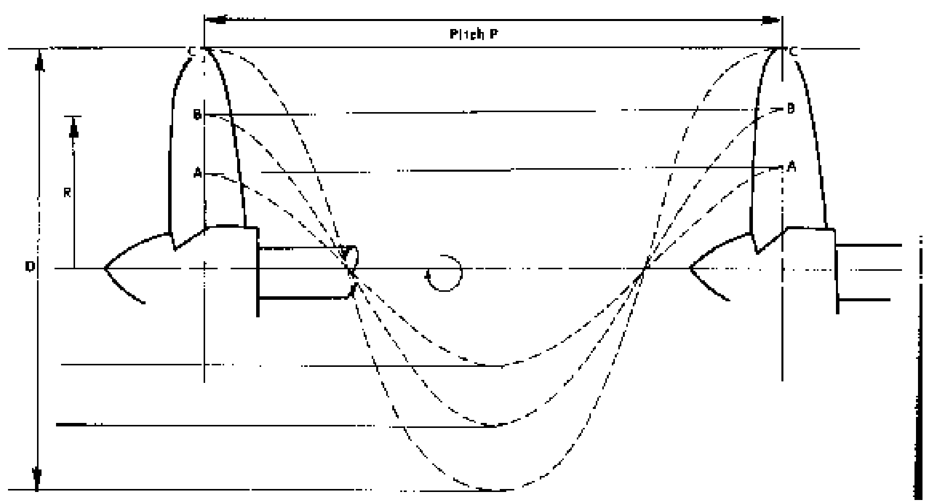


Figure 7.8 - Fixed Pitch Propeller Operating at Zero Slip for One Revolution

Pitch Angle The pitch can also be quantified as a pitch angle which is the angle of the blade from perpendicular to the flow.

Fixed Pitch The pitch is constant all the way from the blade root to the blade tip. Additionally, the pitch of the blades cannot be changed.

Variable Pitch The blades vary in pitch at each radial distance from the hub. This gives the propeller improved efficiency over a wide range of ship speeds.

Controllable Pitch The position of the blades relative to the hub can be changed while the propeller is rotating. This can significantly improve the control and ship handling capabilities of a ship. It also obviates the need for a prime mover reversing mechanism because the blades can provide reverse thrust themselves. Found on FFG-7, DD-963, CG-47, and DDG-51 class ships, for example.

Left Handed Screw Rotates counterclockwise when viewed from astern. Single screw Naval ships use this type.

Right Handed Screw Rotates clockwise when viewed from astern. Twin screw Naval ships use one left and one right handed screw, generally configured to rotate outboard.

Pressure Face High pressure side of the blade. Astern side when moving the ship forward.

Suction Face Low pressure side of the blade. Forward side when moving the ship forward. Most of the pressure difference developed across a propeller blade (and an airplane wing, for that matter) occurs on the low pressure side.

Leading Edge Hits the water first.

Trailing Edge Encounters the water last.

With these definitions known, it is now possible to consider some of the characteristics of propeller operation

7.9.2 The Momentum Theory of Propeller Action

There are several theories of propeller operation, including the momentum theory, impulse theory, blade element theory, and circulation theory. The momentum theory is chosen for discussion here because it gives some valuable insight into the operation of a propeller without the burden of advanced mathematics.

7.9.2.1 Speed of Advance (V_A)

Before a study of momentum theory can proceed, it is required that the concept of speed of advance (V_A) is understood. Figure 7.9 displays the situation.

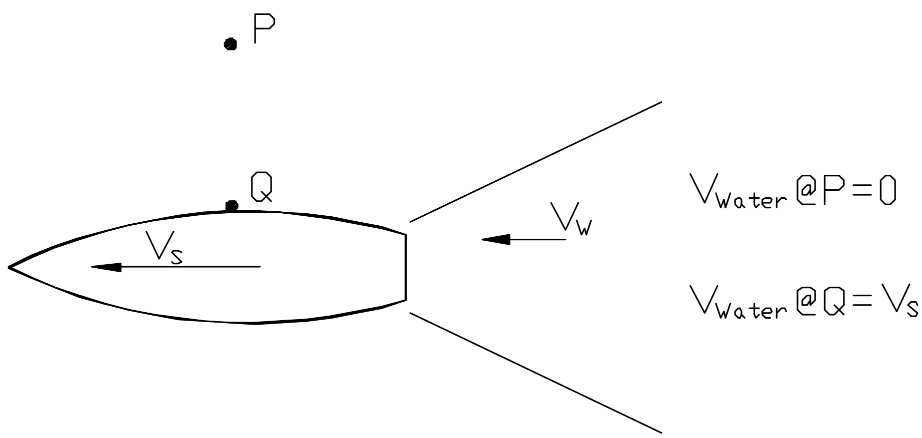


Figure 7.9 - Speed of Advance

As a ship moves through the water, it drags the surrounding water with it. At the stern of the ship, this causes the wake to follow the ship with a wake speed (V_W). Consequently, the screw propeller is experiencing a flow speed less than the ship speed (V_S). The flow speed the propeller is experiencing is called the speed of advance (V_A).

$$V_A = V_S - V_W$$

Example 7.4 What is the speed of advance of a ship traveling at 22 knots that is creating a wake with a wake speed of 3.2 knots?

$$V_A = V_S - V_W = 22.0 \text{ knots} - 3.2 \text{ knots}$$

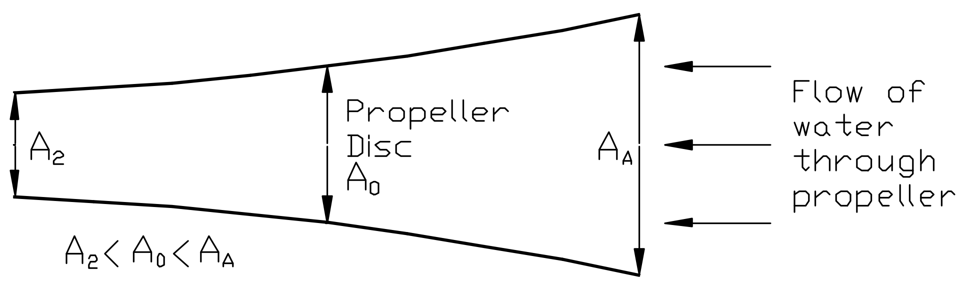
$$V_A = 18.8 \text{ knots}$$

7.9.2.2 Variations in Area, Speed and Pressure Through the Propeller

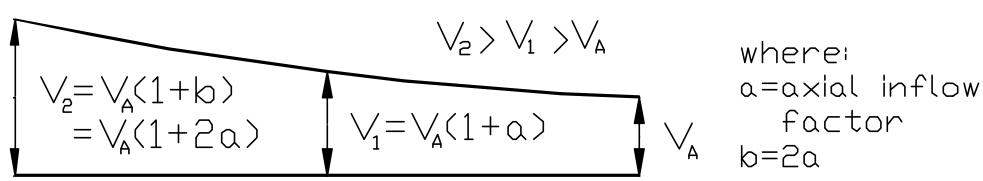
In this theory, the exact nature of the propeller is not important. It is treated as a disc of area A_0. The propeller causes an abrupt change in pressure in the fluid as the blades are crossed, causing the fluid to speed up.

It is important to understand what happens to the cross sectional area, the speed of the fluid, and the pressure variation of the fluid flowing through the propeller. These have been sketched below.

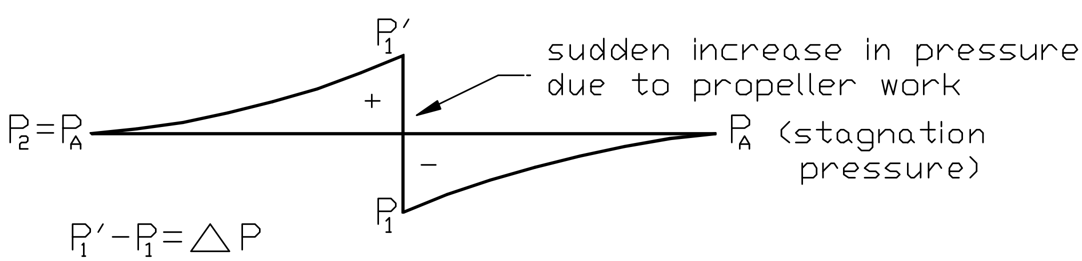
! Area decreases along the control volume



! Speed increases as the area decreases



! Pressure decreases as velocity increases



7.9.2.3 Momentum Theory (OPTIONAL)

Thrust (T)

Consider Bernoulli's Principle Between "A" and "1" - upstream of the propeller

$$P_A + \frac{1}{2}\rho V_A^2 = P_1 + \frac{1}{2}\rho V_1^2 \qquad \text{Equation 1}$$

Consider Bernoulli's Principle Between "2" and "1" - downstream of the propeller

$$P_2 + \frac{1}{2}\rho V_2^2 = P'_1 + \frac{1}{2}\rho V_1^2 \qquad \text{Equation 2}$$

Subtracting Equation 1 from Equation 2 and remembering that $P_2 = P_A$

$$\frac{1}{2}\rho (V_2^2 - V_A^2) = (P'_1 - P_1) = \Delta P \qquad \text{Equation 3}$$

The propeller creates an abrupt change in pressure in the fluid as the blades are crossed. This causes the fluid to speed up. The control volume contracts because the conservation of mass is applicable. This produces Thrust (T).

$$T = (\Delta P)A_0 \qquad \text{Equation 4}$$

Combining Equations 3 & 4

$$T = (\Delta P)A_0 = \frac{1}{2}\rho A_0 (V_2^2 - V_A^2) \qquad \text{Equation 5}$$

Equation 5 represents the change in momentum between points "A" and "2". From an analysis of the change in velocity along the control volume it was found that:

$$V_2 = V_A(1 + b) \qquad \text{Equation 6}$$

where $b = 2a$ and "a" is a quantity called the axial inflow factor.

Combining Equations 5 & 6 reveals this equation for thrust (T).

$$T = \frac{1}{2}\rho A_0 V_A^2[(1 + b)^2 - 1] = \frac{1}{2}\rho A_0 V_A^2(2b + b^2) \qquad \text{Equation 7}$$

Thrust Loading Coefficient (C_t)

The coefficient of thrust loading (C_t) is the dimensionless form of thrust (T) and is defined as follows.

$$C_t = \frac{T}{\tfrac{1}{2}\rho V_A^2 A_0} \qquad \text{Equation 8}$$

Comparing Equation 8 with Equation 7 it is clear that:

$$C_t = 2b + b^2 \qquad \text{Equation 9}$$

Re-arranging Equation 9 reveals

$$1 + C_t = 1 + 2b + b^2 = (1 + b)^2$$

$$\sqrt{1 + C_t} = (1 + b) \qquad \text{Equation 10}$$

Propeller Efficiency ($\eta_{propeller}$)

Propeller efficiency ($\eta_{propeller}$) can be defined as the output compared with the input.

$$\eta_{propeller} = \frac{Output}{Input} = \frac{Available\ Thrust\ Power}{Ideal\ Thrust\ Power} = \frac{TV_A}{TV_1}$$

$$but \quad V_1 = V_A(1 + a) \qquad \text{and} \qquad a = b/2$$

$$\eta_{propeller} = \frac{TV_A}{TV_A(1+a)} = \frac{1}{1+a} = \frac{1}{1+b/2} = \frac{2}{2+b} = \frac{2}{1+(1+b)} \qquad \text{Equation 11}$$

Combining Equation 10 with Equation 11 reveals this expression for propeller efficiency ($\eta_{propeller}$).

$$\eta_{propeller} = \frac{2}{1 + \sqrt{1 + C_t}} \qquad \text{Equation 12}$$

7.9.2.4 Propeller Characteristics

Momentum Theory reveals the following relationship

$$\eta_{propeller} = \frac{2}{1 + \sqrt{1 + C_t}} \qquad \text{where} \qquad C_t = \frac{T}{\frac{1}{2}\rho V_A^2 A_0}$$

For a given thrust (T), the thrust loading coefficient (C_t) increases as the diameter of the propeller (D) and its area (A_0) decreases. This is fairly obvious because there is less foil surface to generate the thrust.

But if C_t increases, propeller efficiency ($\eta_{propeller}$) reduces. Propeller efficiency is lost.

Consequently, propellers are designed to be as large as possible given the available area under the stern and other material strength constraints. This has the effect of "unloading" the propeller and increasing efficiency.

Example 7.5 Use the equation above to calculate the propeller efficiency when the thrust loading coefficient is zero, 1, 2, 3 and 4.

C_t	$\eta_{propeller}$
0	1.0
1	0.827
2	0.732
3	0.667
4	0.618

Example 7.6 Considering a typical propeller efficiency mentioned earlier in this chapter,

what do you consider to be an achievable level of thrust loading coefficient on an operational propeller?

Since a well designed propeller will have a maximum efficiency of about 70%, it would appear from the table above that a minimum practical thrust loading coefficient for a propeller would be about 2.5 .

7.9.3 Propeller Cavitation

Cavitation is the formation and subsequent collapse of vapor bubbles in regions on propeller blades where pressure has fallen below the vapor pressure of water. Cavitation occurs on propellers that are heavily loaded, or are experiencing a high thrust loading coefficient.

7.9.3.1 Types of Cavitation

There are 3 main types of cavitation.

- **Blade Tip** Blade tip cavitation is common because the tips are moving the fastest and therefore experience the greatest dynamic pressure drop.
- **Sheet** Sheet cavitation refers to a large and stable region of cavitation on a propeller, not necessarily covering the entire face of a blade. The suction face of the propeller is susceptible to sheet cavitation because of the low pressures there. Additionally, if the angle of attack of the blade is set incorrectly (on a controllable pitch prop, for instance) it is possible to cause sheet cavitation on the pressure face.
- **Spot** Spot cavitation occurs at sites where there is a scratch or some other imperfection.

7.9.3.2 Consequences of Cavitation

The consequences of cavitation can include the following:

- Reduction in the thrust produced
- Erosion of the propeller blades
- Vibration
- Noise

In the case of a warship, cavitation is to be avoided because the noise it creates can compromise the security of the vessel. Although the Prairie Masker system, used on *Spruance* class destroyers for instance, is highly effective at reducing machinery noise, improper operation of the propeller can create cavitation and compromise the ship.

7.9.3.3 Preventing Cavitation

Several actions can be taken to reduce the likelihood of cavitation occurring.

- **Fouling** The propeller must be kept unfouled and free of nicks and scratches. It must be protected during operation, particularly when operating in shallow water and during approaches. Even a small nick can result in significant spot cavitation and consequential noise levels.

- **Thrust** The throttleman must not attempt to increase rpm and hence the thrust too quickly when accelerating the ship. Cavitation will occur at large thrust loading coefficients. An analysis of the equation for C_t reveals that high propeller thrust (T) at low ship speeds (V_A) increases the thrust loading coefficient that will result in cavitation.

$$C_t = \frac{T}{\frac{1}{2}\rho V_A^2 A_0}$$

  Throttlemen should give V_A sufficient time to keep up with the increase in T. Sudden accelerations and decelerations will cavitate the propeller.

- **Pitch** Operators of ships utilizing controllable pitch propellers must take care that the pitch setting is correct for ship speed and rpm. Detailed guidance for proper propeller operation can be found in the SOP.

- **Depth** Submarines have an additional concern regarding cavitation, namely depth. The depth of the submarine affects the ambient pressure seen by the water at the propeller. When shallow, the hydrostatic pressure is lower and the propeller cavitates at low thrust loadings and rpm's. As the submarine's depth increases, hydrostatic pressure increases, and cavitation is delayed.

7.9.3.4 Propeller Ventilation

Ventilation is a propeller effect often confused with cavitation. If a propeller operates too close to the surface of the water, the localized low pressure created by the propeller blade can suck air under the water and cause effects similar to those mentioned for cavitation.

Ventilation is possible in ships in a very light condition (small draft), in ships experiencing a large negative trim (trim down by the bow) and in ships operating in rough seas.

HOMEWORK CHAPTER 7

Section 7.2

Ship Drive Train

1. a. Draw a simplified picture of a ship's drive train with a prime mover, reduction gears, bearings, seals, struts, and propeller.

   b. Show where the Brake Horsepower, Shaft Horsepower, Delivered Horsepower, and Thrust Horsepower would be measured.

   c. Rank the Horsepowers in part "1.b" from highest in magnitude to lowest.

2. A ship with a drive train arrangement illustrated by Figure 7.1 has the following mechanical efficiencies.

Reduction Gearbox	95%
Bearings/Seals/Struts	98%
Propeller	68%

Calculate the Thrust Horsepower being created by the propeller when the prime mover has a Brake Horsepower of 10,000 HP.

Section 7.3

Effective Horse Power

3. a. What is Effective Horsepower? In your description give its symbol and units.

   b. How is EHP determined in the design of a ship?

Hull Efficiency

4. Briefly explain why is it possible to have a hull efficiency greater than 100%.

Section 7.4

Propulsive Coefficient

5. Towing tank testing has predicted that a ship will have an EHP of 33,000 HP when traveling at 25 knots. What will be the required SHP if the ship has a propulsive coefficient of 0.6?

6. A twin screw ship has the following power data.

Ship Speed (kts)	EHP (Hp)
0	1
6	50
10	110
11	180
12	250
13	360
14	520
15	820

a. **Plot the power curve for this shi Remember to label the axis correc**

b. **Assuming a propulsive coefficien calculate the top speed of the sh it is running on both engines de 700 HP each.**

c. **Assuming the same PC, calculate t of the ship operating on one eng rated at 700 HP.**

Section 7.5

Total Hull Resistance

7. It is common practice to use non-dimensional coefficients when studying and experimenting in the field of fluid dynamics. Briefly describe the advantages of this practice.

8. Give the equation of the non-dimensional form of the Total Resistance of a ship. Briefly describe the terms in the non-dimensional coefficient.

9. What would happen to the total resistance if ship's draft were increased?

Section 7.6

Components of Total Resistance

10. a. Name the components of total hull resistance of a ship in calm water.

    b. Which component dominates at slow speeds?

    c. Which component dominates at high speeds?

Viscous Resistance

11. a. Briefly describe the 2 components that make up the Coeff Resistance experienced by a ship moving through calm wa

b. Referring to the empirical equations for these compone happen to the Coefficient of Viscous Resistance if the kin of the water increased?

12. a. Define laminar and turbulent flow.

    b. Draw an aerial (looking from the top down onto the weatherdeck) view of a moving ship showing laminar flow at the bow, the transition point, flow separation, and turbulent wake.

    c. What is done to a model to ensure it experiences turbulent flow as it moves through the towing tanks in the laboratory?

13. What measures can be taken to reduce the Coefficient of Viscous Resistance of a ship?

Wave Making Resistance

14. Briefly describe the 2 major wave systems generated by a ship moving through calm water. Use a sketch in your description.

15. Why are there humps and hollows in a ship's resistance curve? Sketch and label this curve.

16. What measures can be taken to reduce the Coefficient of Wave Making Resistance of a ship?

Other Types of Resistance

17. Name and explain four other types of resistance not included in the total hull resistance.

18. Why does it take more power to do the same speed in shallow water than in deep water?

Section 7.7

Basic Ship Modeling Theory

19. Briefly explain the terms geometric similarity and dynamic similarity.

20. Briefly explain how geometric similarity and partial dynamic similarity is achieved in ship resistance testing.

21. An FFG-7 (L_{PP} = 408 ft) is being modeled at 28 kts. The scaling factor is 12.

    a. Calculate the length of the model.

    b. Calculate the speed of the model if the Coefficient of Wave Making Resistance is to be the same for the ship and the model. [1 knot = 1.688 ft/s]

Section 7.8

Model Testing and EHP Calculation

22. Model testing for a DD-963 was conducted. The results were as follows:

	Model	**Ship**
Length, L (ft)	6.625	530
Beam, B (ft)		55
Draft, T (ft)		19.5
Displacement, Ä (LT)		7800
Surface Area, S (ft^2)		33,150
Density of water, ñ ($lb\text{-}s^2/ft^4$)	1.936	1.9905
Kinematic Viscosity, í (ft^2/s)	1.092×10^{-5}	1.2791×10^{-5}
Correlation Allowance		0.0003
Speed, V (ft/s)	4.2	
Total Hull Resistance, R_T (lb)	0.45	

Conversion factor: 1 knot = 1.688 ft/s.

Using the information given above to find the following:

a. Scale Factor

b. Wetted surface area of the model

c. Displacement of the model

d. Corresponding Speed of the Ship in knots

e. Reynolds number of the model

f. Form factor

g. Coefficient of friction of the model

h. Coefficient of viscous resistance of the model

i. Coefficient of total hull resistance of the model

j Coefficient of wave making resistance of the model

k Coefficient of wave making resistance of the ship

l. Reynolds number for the ship

m. Coefficient of friction of the ship

n. Coefficient of viscous resistance of the ship

o. Coefficient of total hull resistance of the ship

p. Total Hull Resistance of the ship in lbs

q. EHP of the ship

r. SHP of the ship if the propulsive coefficient is 60%.

23. In the previous problem you were required to calculate the displacement of the model being tested. Why is this calculation important to model testing?

Section 7.9

Screw Propeller Definitions

24. On a sketch of a screw propeller, show the hub, blade tip, blade root and the propeller diameter.

25. Describe 2 methods of quantifying the pitch of a propeller.

26. Briefly describe the differences between, fixed pitch, variable pitch and controllable pitch propellers.

Coefficient of Thrust Loading

27. Using the equations in the text for Coefficient of Thrust and Ideal Propeller Efficiency, answer the following:

    a. Will a larger propeller be more or less efficient than a small one?

    b. Will high thrust and low ship speed give high or low propeller efficiency?

Cavitation

28. Briefly describe why propeller cavitation occurs.

29. What is the relationship between thrust loading and propeller cavitation?

30. Explain the following terms:

    a. Blade Tip Cavitation

    b. Spot Cavitation

    c. Sheet Cavitation

31. What measures can an operator take to minimize propeller cavitation?